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EXPERIMENTAL AND NUMERICAL INVESTIGATION FOR THE NATURAL CONVECTION HEAT TRANSFER IN AN ENCLOSURE HAVING BAFFLES

Nabil Jamil Yasin
Engineering Technical Collage, Middle Technical University, Baghdad, Iraq

Dhia Al-Deen H. Alwan
Engineering Technical Collage, Middle Technical University, Baghdad, Iraq

ABSTRACT

In this paper, the natural convection heat transfer in a cubic enclosure provided with inclined baffles attached to the two adiabatic sides, heated from the bottom is studied experimentally and numerically to assess the effect of the baffles on the heat transfer process inside the enclosure. Two different configurations have been considered. The first configuration corresponds to the heated from the bottom with uniform heat flux using two baffles attached to the left and right walls, while the second configuration corresponding that the enclosure's floor has parallel bands that are heated to a constant, high temperature and the bands are separated by gaps that are kept at a lower temperature that is also constant and single baffle attached to the left wall. In both cases, the top wall is kept at a lower temperature than the bottom wall and the inclined baffles are well covered with an insulating material. The inclination angles of the baffles range as ($0^o \leq$ and $\geq 150^o$). The governing parameter, Rayleigh number, is fixed within $2.6x10^{11}$. In numerical solution, a commercial software package has been used for a 2-D computation, and the effect of turbulence is modelled by using (k-ε) model. Depending on its orientation, the partial baffle has been found to change significantly the flow field which in turn causes a reduction to the heat exchange inside the enclosure due to the damping caused to the flow field. For all cases, the insulated baffle with any inclination angle caused a reduction to the heat exchange inside the enclosure due to the damping caused to the flow field. Also, a good agreement has been obtained between experimental measurements and numerical results.

Keywords: natural convection, turbulence, enclosure, inclined partial baffles.

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1. INTRODUCTION

Natural convection heat transfer in differentially heated baffled cavities are used in a various industrial application such as heating and ventilation of a living space, fire in building solar thermal collector system. The baffles are added to improve and control the heat transfer and fluid flow characteristics. Studies of various aspects of this problem have been carried out by many researchers both theoretically and experimentally. Some studies focused on the natural convection inside enclosure motivated by heating and cooling the horizontal walls, while the vertical walls are insulated. While others discussed natural convection in an enclosure heated and cooled through the vertical walls while the horizontal walls are kept adiabatic, the matter which has received a great consideration in studies, that is due to much industrial application use these concepts. Nansteel and Greif [1] studied the convection heat transfer process and fluid flow occurring in the 2D rectangular enclosure fitted with partial vertical divisions. The horizontal walls of the enclosure were adiabatic, while the vertical walls were maintained at different temperatures. The effect of the baffles on the heat transfer across the enclosure was determined and a correlation for the Nusselt number as a function of Rayleigh and baffle lengths were generated for both conducting and non-conducting baffle materials. Bajorek and Llyod [2] investigated the natural convection heat transfer within a baffled enclosure of aspect ratio (1). The vertical walls were maintained isothermally at different temperatures, while the horizontal walls and the baffles were insulated. They found that the baffles significantly influenced the heat transfer rate. Nansteel and Greif [3] investigated the effect of the baffle or orientation on the heat transfer and fluid flow in a rectangular enclosure fitted with a vertical adiabatic baffle. The baffle was oriented parallel to the vertical isothermal walls, one of which was heated and the other cooled while all other surfaces of the enclosure were insulated. The effect of the transverse location was examined and reported. Frederick [4] studied natural convection in an air-filled, differentially heated, inclined square cavity, with a diathermic baffle on its cold wall numerically. The baffle cause convection suppression and heat transfer reduction up to 47% relative to the undivided cavity at the same Rayleigh number, baffle length, and inclination. Neymark et. al. [5] studied the effect of internal baffles on the flow and heat transfer characteristics of air and water filled partially divided enclosures at high flux Rayleigh number. Experiments were conducted using a representative cubic geometry differentially heated from the side with an internal partial vertical baffle. The study showed that the Nusselt number became a strong function of aperture width, and the temperature difference across the aperture approached the overall enclosure temperature difference. Ambarita. et al. [6]were studied a differentially heated square cavity, formed from two horizontal adiabatic walls and two vertical isothermal walls, with two perfectly insulated baffles attached to its horizontal walls numerically. It was observed that the two baffles trap some fluid in the cavity and affected the flow fields. Also, it was found that Nusselt Number is an increasing function of Rayleigh number, a decreasing one of baffle length, and strongly depends on baffle position. Ghassemi et. al. [7] investigated the effect of two insulated horizontal baffles placed at the walls of a differentially heated square cavity numerically. The vertical walls are maintained at different temperatures while the horizontal walls are adiabatic. The result shows that the two baffles trap some fluid in the cavity and affect the flow. Asif, et al [8] were carried out a numerical study to investigate the mixed convective two-dimensional flows in a vertical enclosure with heated baffles on side walls. All walls are assumed to be adiabatic, but baffles are considered as isothermally heated. Heated baffles are placed both at the left and right wall of the enclosure. It was observed that maximum heating efficiency is found at a higher value of Reynolds and Richardson number. Abid [9] studied the natural convection of an air-filled partitioned rectangular enclosure numerically. Top and bottom of the enclosure were adiabatic; the two vertical walls are isothermal. Two perfectly insulated baffles were attached to its horizontal walls at a symmetric position. The results of the values of average Nusselt number and maximum absolute stream function have been confirmed by comparing it with similar previous

works using the same boundary conditions and a good agreement was obtained. Mushatet[10] investigated the laminar natural convection inside a rectangular cavity containing two cylindrical obstacles numerically. The cavity was differentially heated. The governing partial differential equations are solved using stream function and vorticity method. The effect of the distance between the obstacles has been tested. The results show that the fluid flow and temperature fields significantly depend on the distance between the obstacles for the studied Rayleigh numbers. Mushatet [11] investigated the turbulent natural convection heat transfer and fluid flow inside a square enclosure having two conducting solid baffles numerically. Fully elliptic Navier-Stokes and energy equations are discretized using finite volume method along with staggered grid techniques. The results show that the rate of heat transfer is increased with the increase of Rayleigh number especially for the region near the baffles. Varol, et. al. [12] studied experimentally and numerically the natural convection heat transfer in an adiabatic inclined one-fin attached one side of the square enclosure. The bottom wall of the enclosure has a higher temperature than that of the top wall while vertical walls are adiabatic. It was observed that the inclination angle affects the flow strength and temperature distribution. Enayati, et.al [13] carried out a 3-D large eddy simulations (LES) of natural convection in a laterally heated cylindrical reactor. The objective was to understand the effect of the opening area of the baffle on the flow pattern and temperature distribution inside the reactor. The baffles considered in this study are annular hollow discs with different opening areas. Velocity and temperature distributions across the different planes and lines are analyzed in order to obtain information on the flow and heat transfer processes resulting from various baffle openings. Pushpa, et.al [14] examines the influence of a circular thin baffle on the convection in a vertical annular enclosure. The inner and outer cylindrical walls and the baffle are retained with different temperatures and concentrations, while the upper and lower boundaries are kept at adiabatic and impermeable. It has been observed that the baffle size and location has a very important role in controlling the convective flow and the corresponding heat and mass transport characteristics.

In this work, experiments and computation are conducted to investigate the effects of adiabatic partition and its inclination on the natural convection heat transfer in a cubic enclosure. Two configurations were considered. In the first configuration, two baffles were fixed to the adiabatic side's walls of the enclosure and the bottom wall was heated with constant heat flux, while in the second configuration, single baffle was attached to the left wall of the enclosure while the bottom wall was heated with separated, parallel high-temperature bands to mimic rows of heated equipment. For both cases the top wall was maintain a constant lower temperature and the baffles can vary its orientation with respect to the horizontal side of the enclosure. The present work aim to show how the angle of the inclination can affect the flow and thermal field characteristics of the inclined baffled enclosure in the turbulent natural convection under the above boundary conditions.

2. SIMULATION SETUP

The CFD software Fluent-ANSYS15, with 8,200 finite-volume-method cells were used for the simulation of the steady-flow RANS scheme with the standard k-ε turbulence model and standard wall functions; and governing equations are those associated with turbulent natural convection. The boundary condition is: Non-slip condition on all surfaces (U=0, V=0), the enclosure filled with a fluid of Prandtl number (Pr = 0.725). The fluids properties correspond to those of air are assumed to be constant; but Boussinesq approximation applies for the temperature-induced change in density giving rise to buoyancy. The schematic of the Physical situation of two configurations under study are shown in figures (1) and (2) which is a middle section of a cubic enclosure with a perfectly insulated vertical walls to be kept in adiabatic conditions. The baffle with length (B) and thickness of (t) located in the middle of the vertical walls and inclined with inclination angles ranged ($0^o \leq$ Angle $\leq 150^o$) as shown in the figures. The top wall of the

enclosure is kept at constant temperature. For case (1), the bottom hot wall keeps at uniform heat flux. Two baffles with different inclination angles denoted by (θ), and (β) are used for the analysis of this case. The bottom wall of the enclosure was keeps at uniform heat flux as shown in Fig.(1). A high Rayleigh number (Ra=2.6x1011) was considered during investigation. For case (2), the bottom hot wall exposed to step function heating with alternating temperatures of 358K and 381K respectively, in 12 steps as shown in Fig.(2). The top wall is isothermal at 275K. Rayleigh number value is 10^9 indicating that the buoyancy-induced flow inside the enclosure is turbulent. All other boundary conditions are shown in the mentioned figures.

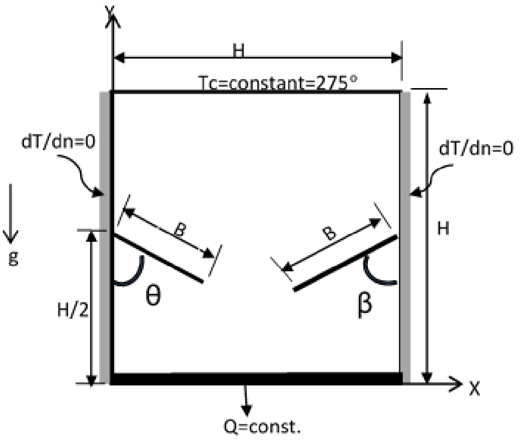

Figure (1): Schematic drawing for the enclosure corresponding to case (1)

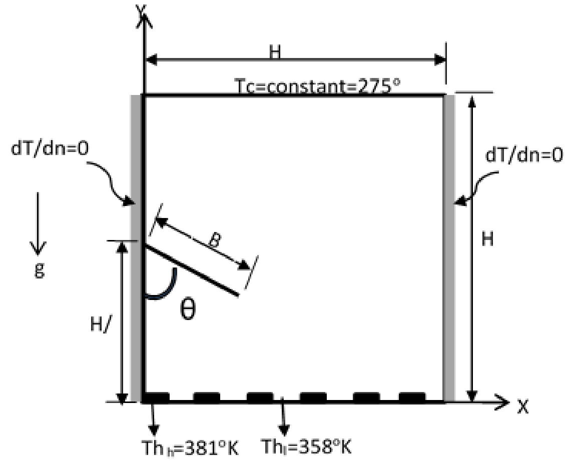

Figure (2): Schematic drawing for the enclosure corresponding to case (2)

3. EXPERIMENTAL SETUP

Figure (3) shows the experimental apparatus which used for this work. The main part of the test rig is a cubic enclosure (30x30x30cm) which constructed from three sides by the use of low conductivity block wood, and from the front side by the use of double pan glass window to allow visualize. The top side is constructed from pure aluminium sheet (0.1mm) fabricated as fully closed container with gate to use as crashing ice-vessel, and covered from all outsides by (25mm thickness) styrol-board to insure constant temperature of the top wall at about 2ºC ±0.5ºC. A drain pipe is connected to the bottom of the crashing ice-vessel to drain molten ice water continually and prevents the forming of high temperature film of water beneath the crashing ice. After steady state condition reached in the enclosure, the amount of molten ice water is collected

in a scaled beaker for 10 minutes, to calculate the amount of heat received from the enclosure by the ice to melt in this period of time. The thermodynamic heat balance equation

$$Q_{icemelt} = \dot{m} \cdot h_{fg} \tag{1}$$

Was used to determine the amount of heat transfer from the top side. This can be used as an indicator to the accuracy of the insulation. For both cases, as mentioned before, the adiabatic partial partition with length B=0.135m was located in the middle of the vertical walls and inclined with different inclination angles denoted by θ, ($0^o \leq \theta \leq 180^o$) were used. The non-heat conductive partial partition was fabricated from thin metal coated from the two sides with rubber with overall thickness (t=0.005m).The enclosure was filled by air with prandtl number of 0.725.

For the case (1) configuration, the bottom wall of the enclosure is constructed from sheet of pure aluminum (8mm) thickness to insure better uniform heat flux. An electrical resistance heater fixed under the aluminum sheet, constructed from strips of (1 mm) width made of chrome-nickel alloy with resistance (10 ohm/m) and wrapped with (5 mm) pitch around a (300x300 mm^2) mica sheet of (0.5 mm) thickness, to ensure the electrical insulation. The overall resistance of the heater used is (96 ohm). The heater is covered from bellow by a glass wool of (80 mm) thickness to reduce the heat loss and then covered with a wood cover plate. Eight thermocouples (0.3mm) copper-constantan type-T distributed in two horizontal plane level parallel to the plane of the heater (four in each plane) are plant in equal distances from all sides inside the glass wool. The first thermocouples plane located (5mm) from the heater plane whereas the second one located (60 mm) apart, for the purpose of heat loss measurement from the bottom side of the heater. The thermocouples were calibrated to measure a temperature difference error of about (±0.5^{o}C).To produce a step function of temperatures distribution for configuration (2), an uncoated and coated strips formed by the used of low conductivity coating material which allows to get the step temperatures distribution in 12 steps of 381K and 358K respectively as a result of increasing the thermal resistance in the coated strips. Two guard heaters produced in the same manner of the main heater (ring heater 55 ohm, and base heater resistance 65ohm) were used to prevent heat transfer in the lateral and downward directions. The heaters are covered from the sides and from bellow by a glass wool of (80 mm) thickness to reduce the heat loss. Nine thermocouples were interplant in the main Aluminum plate to insure a constant temperature distribution in the plate, and eight were interplant in the two guard heaters four in each one for the purpose of controlling the electrical current that must be flow throw each heater to insure a constant temperature in the heating plate. The thermocouples were calibrated to measure a temperature difference error of about (±0.3^{o}C). The heaters were supplied with AC-current via five voltage regulators. The voltage and current supplied to each heater was measured with a calibrated volt meters and ampere meters at an accuracy of about (±1.2%). The net power supplied to the main heater as a source of heat enters to the enclosure was (380 W/m^2). This power represented the electrical power measured from the reading of the volt and ampere meters. For the purpose of measuring the local heat transfer coefficient above each strip in the direction perpendicular to the partial partition orientation, the temperatures inside the enclosure in three locations (strip surface, 0.2mm & 0.4mm above the hot wall surface) of the 12 strips were measured using a sheathed thermocouple probe type(TD745) with accuracy of (±0.2^{o}C). The sheathed thermocouple probe could be move parallel to the hot strips from a small hole in the side of the enclosure.

For both cases, the heater is carefully mounted on a (320x320 mm^2) mica sheet and covered with other (320x320 mm^2) mica sheet to prevent the electrical contact. All heaters was bounded with wood box (25mm thickness) to prevent any heat loss from bellow and from the lateral sides. The adiabatic partial partition with length B=0.135m was located in the middle of the vertical walls and inclined with different inclination angles denoted by θ, ($0^o \leq \theta \leq 180^o$) were used. The

non-heat conductive partial partition was fabricated from thin metal coated from the two sides with rubber with overall thickness (t=0.005m).

Figure (3). Schematic diagram of experimental set-up.

3.1 Data reduction

To calculate the local heat transfer coefficient and then the local Nusselt number in the middle line of the hot wall plane, perpendicular to the baffles orientation, the hot wall surface divided into 12 part, and a 12 thermocouples type-T were plant in the hot wall only 0.2mm from the hot surface, insert from the bottom of the hot wall. The air temperatures inside the enclosure in a location (0.2& 0.4mm) above the hot wall surface in the centers of the 12 parts were measured using a sheathed thermocouple probe type(TD745) with accuracy of (±0.2°C) which could be moved parallel to the hot wall enter from a small holes in the side of the enclosure. This air temperatures measurements allows find he temperature gradient in each one of the 12 parts of the hot wall and then can find the local and average Nusselt number, according to the following equations [15]:

$$Nu_x = -\frac{\left.\frac{dT}{dy}\right|_{y=0} . H}{T_h - T_c} = \frac{q'' H}{k(T_{h_x} - T_c)} \tag{2}$$

And the average Nusselt number can be calculated according to:

$$\overline{Nu} = \int_0^H Nu_x dx \tag{3}$$

Rayleigh number can be calculated according to:

$$Ra = \frac{g\beta H^4 q''}{k\upsilon\alpha} \tag{4}$$

The experimental test repeated three times for each case of partition inclination angle. Each experimental test required at least 90 minutes to reach study state condition. The steady state

condition materializes when the 9 thermocouples interplant in the heating Aluminum plate measured the same temperature measured in the 4 thermocouples interplant in the ring guard heater and the 4 thermocouples interplant in the base guard heater. The thermal physical properties of the air are measured according to the bulk mean temperature of the average temperature of the heating wall and the cooling wall temperature. According to the uncertainty analysis given by Holman [17], Nusselt number uncertainty analysis shows that the maximum error is (±0.552). Repeatability check details of the temperature distribution measured for three times repeated tests for each run tests, show that the percentage difference in the readings is not exceed 5% for all readings.

4. RESULTS AND DISCUSSION

In this investigation an experimental and numerical solutions is done to the problem represented by the effect of change the inclination angles for the adiabatic baffles attached to the vertical adiabatic walls. The results represent the local Nusselt number with respect to x-position, the velocity field and temperature distribution contours for both the cases. The numerical results represent the local and average Nusselt numbers which compared with the experimental results to validate the work. Also the flow velocity field and temperature distribution counters for each case under investigation are got and shown.

Case (1)

Figure (4) shows the collection of (β=0) and (θ) changes from 0^o to 150^o. The left column shows the local Nusselt number with (x=0→0.3m) experimentally and numerically which shows that local Nusselt number increase with (x) to some value and then decrease depends on the baffle inclination angles, and upon the direction of the cell circulating. Generally, Nusselt number decrease as (θ) increase up to 90^o and then it will increase slightly up to (θ =150^o). The flow field contour at the middle column and the temperature distribution column right column explain this situation. When (θ =0), the whole of the enclosure act to transfer the heat from the bottom wall to the top wall but, when (θ) increase a part of the enclosure act as heat trap, and this trap increase as (θ = 90^o), and two circulating cells start to form, one of a hot air near the hot wall and other cold near the cold wall with little mixing between them at the centre of the enclosure. As (θ) increase farther up, main circulating cell will form with another small one which results increasing the local Nusselt number. Figure (5) shows the collection of (β=60) and (θ) changes from 0^o to 150^o in the same manner as before. The maximum local Nusselt number shift more to the left due to the effect of right baffle, and a two cells forms earlier as (θ =0^o). When (θ =30^o) the lower cell shown to be contracted and divided into two cells in the vertical direction, and then the two cells divided into six cells with small sub-cells as (θ =60^o) in a symmetry shape. When (θ =90^o) the upper half forms one main cell but the lower half of the enclosure forms three main cells with many sub-cells in the two half's. The cells start to form inverse S-shape for (θ =120^o) with some sub-cells and the number of cells reduces as (θ =150^o). The multi cells forms reduce the heat exchange between the hot & cold walls, and that is explain the reason of reduce the level of local Nusselt number in this case. The maximum local Nusselt number noticed when (θ =0^o) and then when (θ =150^o).Figure (6) shows the collection of (θ =150^o) and (β) changes from 0^o to 150^o. The large cells in this collection is the reason of relative high local Nusselt number, but the difficulty of the fluid flow in the cases of (β=60^o, 90^o, 120^o) cases the reduction of it. Figure (7) shows the local Nusselt number with x-position on the left of each sub-figure and average Nusselt number with inclination angles on the right of each sub-figure from (1) to (4) as a compact of each collection which are discussed. The right figures show clearly that the average Nusselt number decrease as the inclination angles of the baffles increase for all collections and then increase. For all cases, the long insulated baffle of any inclination angle causes a reduction to the heat exchange inside the enclosure due to the damping cause to the flow field.

Case (2)

Figure (8) show the contours of velocity magnitude (m/s) for the enclosure with and without partial partition with different inclination angle. For the purpose of illustration, the bottom of the box will be divided into (12) strips so that the number (1) represents the first left strip as shown in the top of the figure. Part (A) represent the enclosure without partition. Inspection of the figure shows that the space above strip number (1) has a small revise circulation portion in the corner of the enclosure causes a reduction in the heat exchange, whereas the space above strip number (12) shows stagnation causes lower heat exchange in compare with all other strips. Moreover, it seems that maximum velocity occurs in the strips number 5 and 6 that is explain the high exchange of heat at that strips. Due to the circulation of flow anticlockwise the upstream of the flow in the first strips (1-6) causes a higher heat exchange compare with the strips (7-12). Part (B) represents the enclosure with a partition inclined by 45^o. The partition causes an obstruction between the hot wall and the cold wall of the enclosure to the strips (1-5) which causes reduce the heat exchange in these strips. Maximum velocity can be noticed over the strips (6). Strips (7-10) also show high level of heat transfer due to the effect of the anticlockwise circulation of the air flow. The left corner under the partition shows a revers flow cause reduce in the heat exchange. Part (C) represents the enclosure with a partition inclined by 90^o.This case the shows that there is no direct contact in the circulation of air in the lower part with the cold surface of the enclosure, which results in reduce the heat exchange compare with the previous cases that is because the enclosure is divided in to two flow field, the upper part circulate anticlockwise whereas the lower part circulate clockwise. The higher flow velocity noticed in strips (7-9). That is explained the reason for the high level of heat exchange shifted to the right side of the enclosure compare with previous cases. Part (D) represents the enclosure with a partition inclined by 135^o. In this case the main flow field circulates clockwise and the maximum flow velocity noticed above strips number (5-7) and a film of Simi stagnation air formed in the upper portion of the enclosure cases a kind of insulation between the main flow field and the cooling surface, causes a reduction in heat exchange inside the enclosure.

Figure (9) shows numerically the contours of Static Temperature (K) for the enclosure with and without partial partition with different inclination angles. Part (A) represent the enclosure without partition, which shows that cold flow field (represented by green color) is come very close to the bottom surface of the enclosure causes effective cooling to the hot wall. Part (B) represent the enclosure with partition inclined by 45^o shows an ineffective cooling field of yellow color in the left side of the enclosure, whereas in part (C) of the 90^o inclination angle and part (D) of the 135^o inclination angle show that the ineffective field of yellow color includes whole the bottom side of the enclosure which reduce the heat exchange.

Figure (10) shows numerically the heat flux on the heating wall under different inclination angles of the partial partition. At (0^o) a high level of heat exchange between the heating surface and the air flow in the enclosure was observed. The highest amount of heat flux with a maximum heat flux about 1200 W/m^{2}occurs in the left strips which represents the leading edge of the flow with respect to the hot wall and reduce gradually at the trial edge. At 45^o a reduction in the heat flux to maximum of 900 W/m^2 reduces to about 700 as a second maximum value was noticed. At (90^o) a maximum heat flux of 800 W/m^2 was near the right side. Figure (11) shows a comparison between the experimental results depends on the experimental tests and the numerical result. The experimental local Nusselt number calculated according to equation (2) which shows that the change of local Nusselt number with x-position experimentally and numerically have good match.

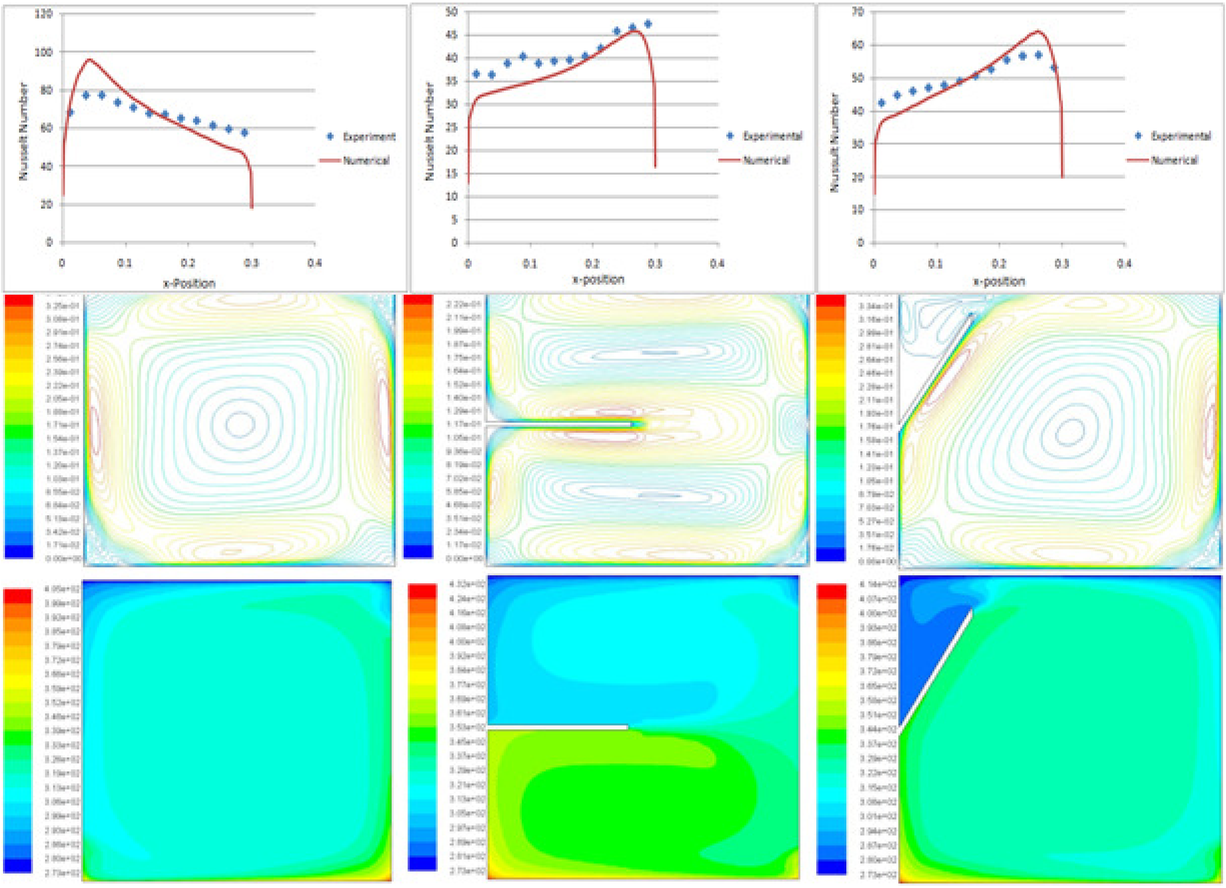

Figure (4) Local Nusselt (upper row), contours of the velocity magnitude (m/s) on the (middle row), and contours of the temperature (K) (lower row), (θ=0, β=0), (θ=90, β=0), and (θ=150, β=0)

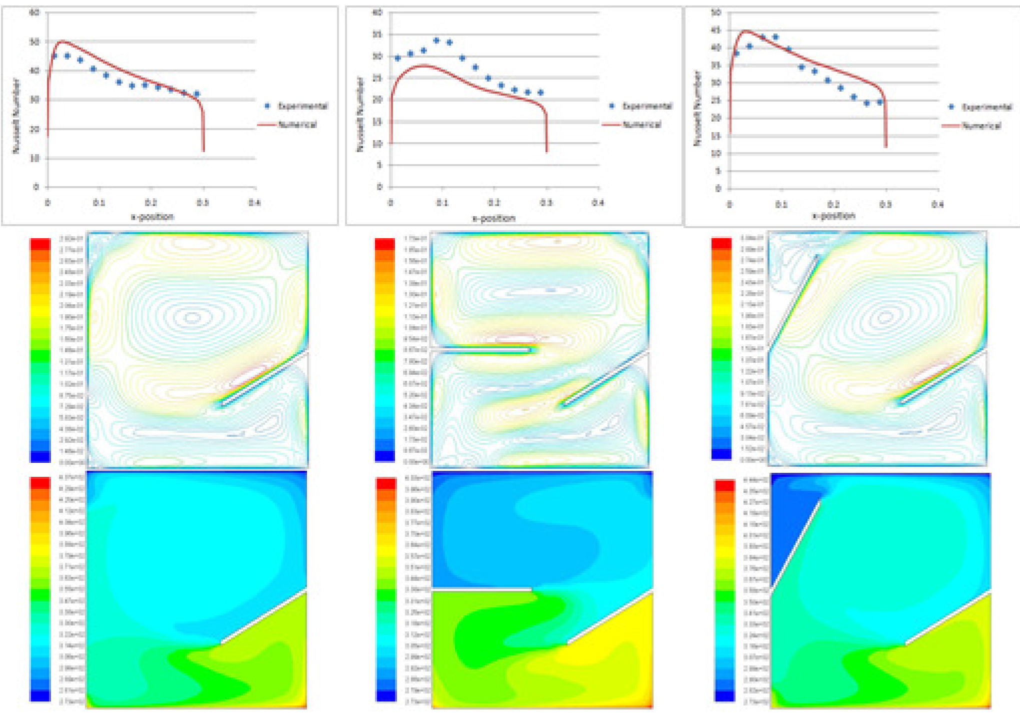

Figure (5) Local Nusselt (upper row), contours of the velocity magnitude (m/s) on the (middle row), and contours of the temperature (K) (lower row), (θ=0, β =0), (θ=90, β=0), and (θ=150, β=0)

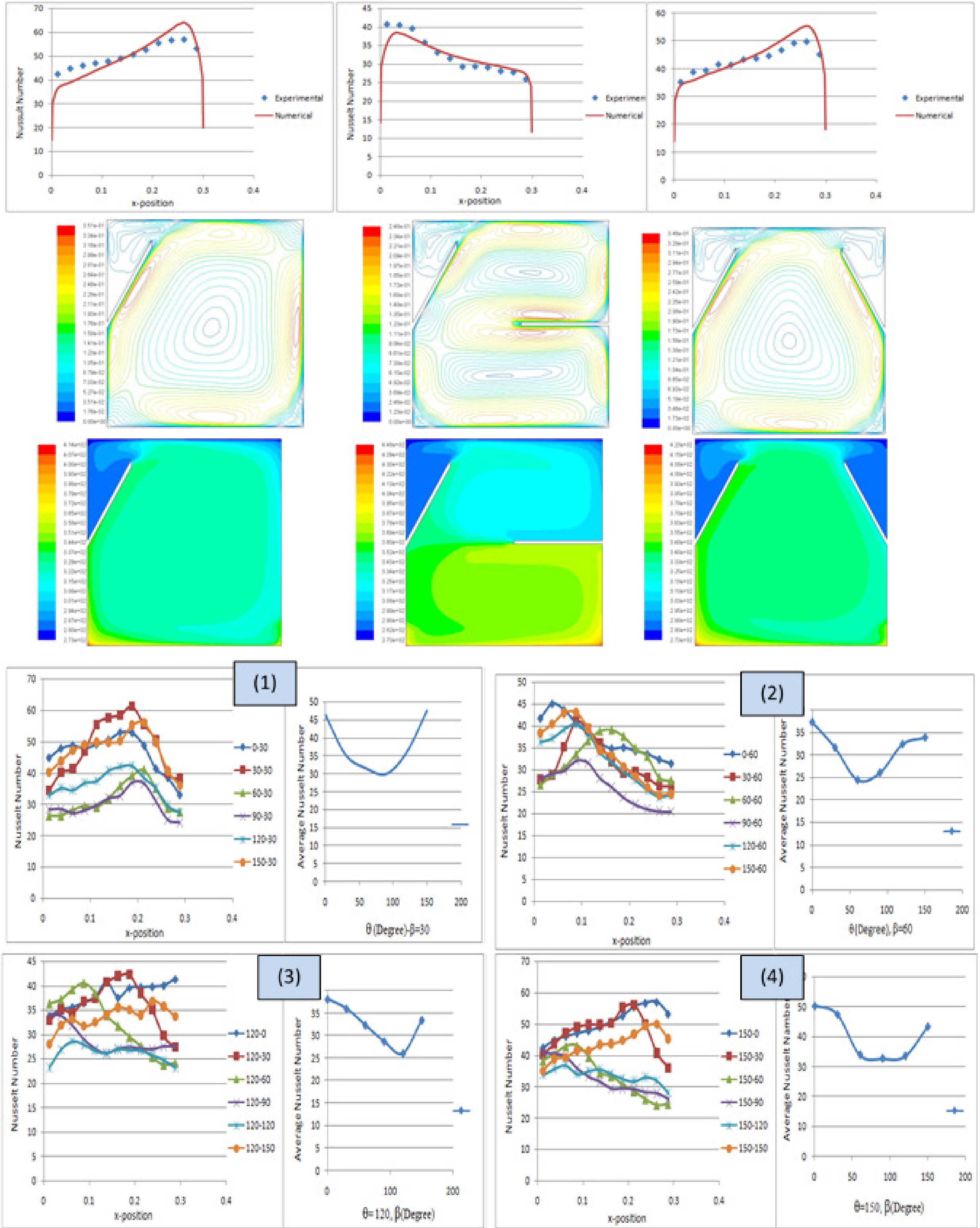

Figure (7) Variation of local Nusselt Number with the x-distance for different baffles orientation θ and β.

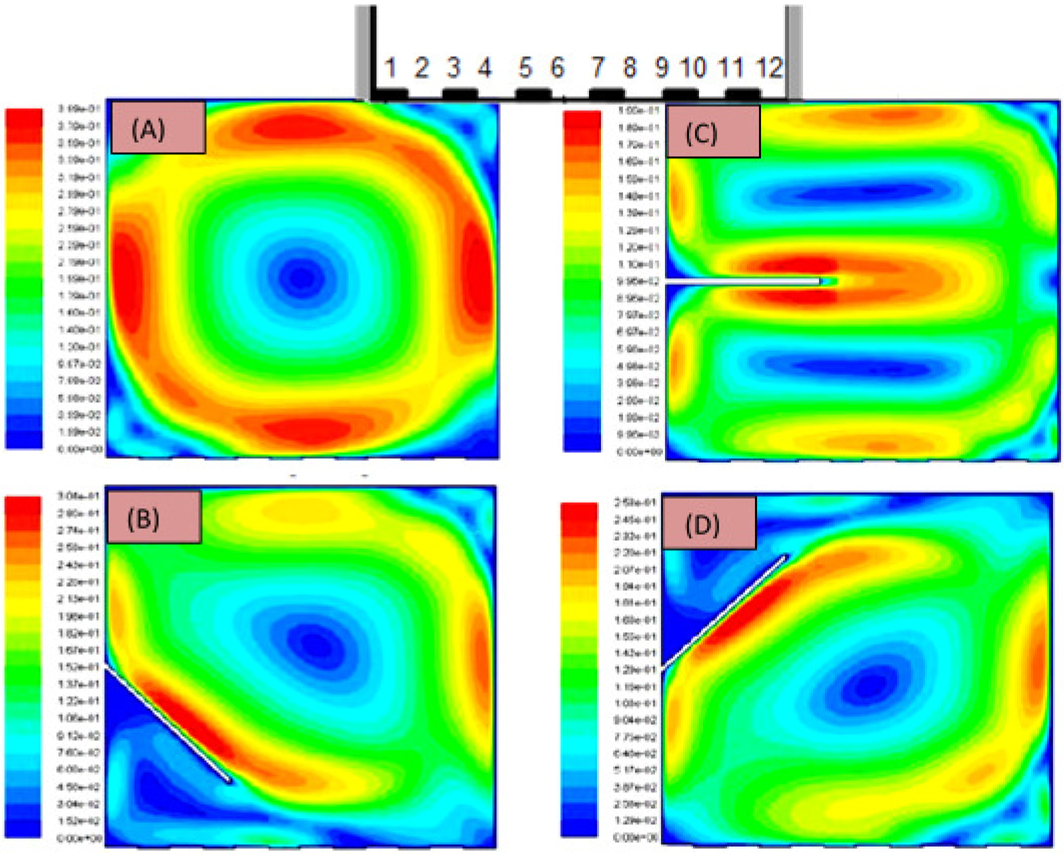

Figure (8) Contours of velocity magnitude (m/s) for the enclosure without partial partition and with partial partition with different inclination angle.

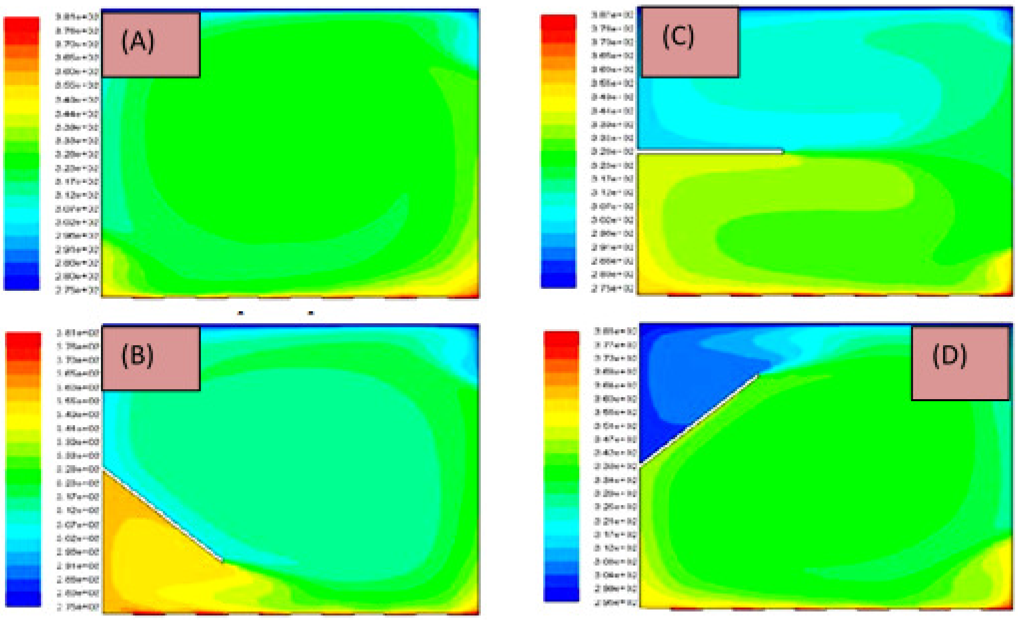

Figure (9) Contours of static temperature (K) for the enclosure without partial partition and with partial partition with different inclination angle.

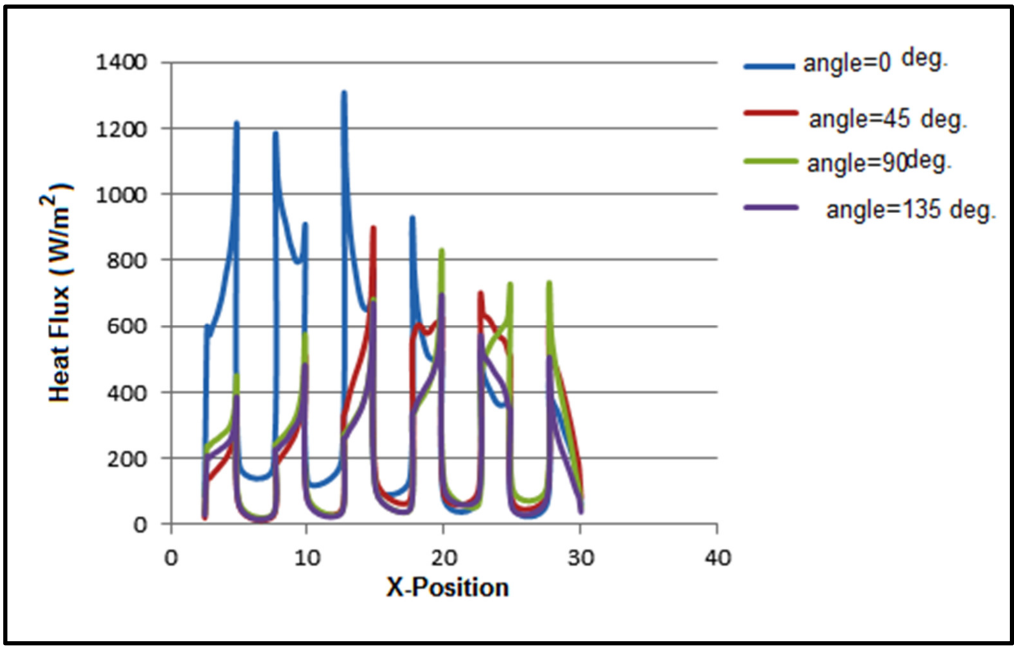

Figure (10) Heat Flux on the heating wall under different inclination angles of the partial partition based on Numerical results.

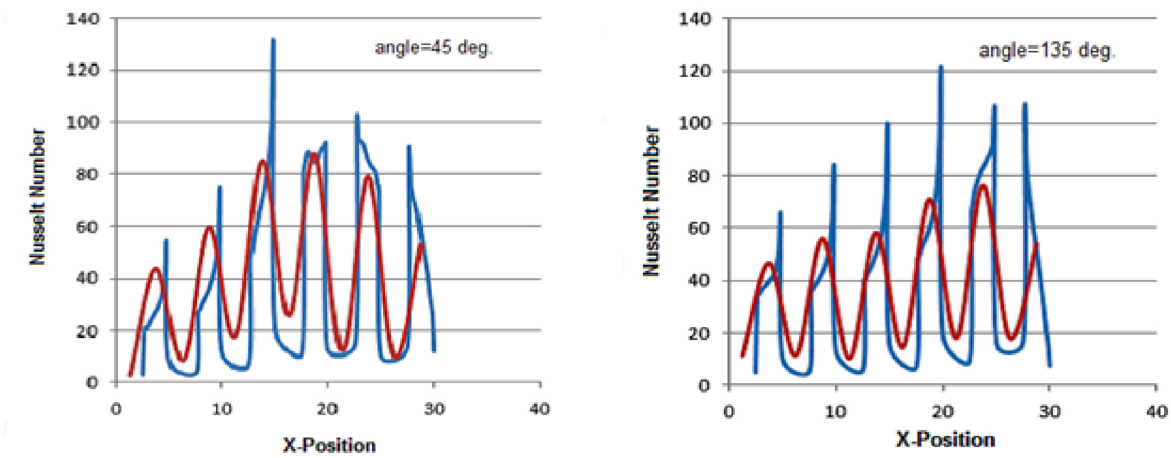

Figure (11) Comparison between the experimental and numerical results for the (Nu) values

(— Num, — Exp.).

5. CONCLUSION

In this investigation, the effect of attached an insulated baffles, oriented with couples of inclination angles, in a facing sides of an cubic enclosure cold from the top by constant temperature heated from the bottom with uniform heat flux or to a step function of high temperatures. And, whereas other sides kept insulated, which has done experimentally and numerically.The following is the important high marks in this investigation:

1. The experimental solution gets a good agreement with the numerical solution to the problem.
2. The long insulated baffle of any inclination angle causes a reduction to the heat exchange inside the enclosure due to the damping cause to the flow field.

3. Increasing the inclination angle of the partition allow increasing the cells inside the enclosure.
4. The average Nusselt number decrease as the inclination angles of the baffles increase for all collections and then increase.
5. The inclined partial partition can be assumed as damping mean to the flow field velocity in a control manner help to keep the temperature of different power out equipment's in the same temperature.

It is recommended that further study to find an empirical equation to determine the average and local Nusselt number as a function of Rayleigh number, (θ), and (β) angles.

NOMENCLATURES

B	Baffle length	(m)	Ra	Rayleigh number	
g	Gravitational acceleration	(m/s^2)	t	Thickness	(m)
h_{fg}	Heat of fusion of water	(kJ/kg)	T	Temperature	(°C)
H	Cubic enclosure length	(m)			
k	Thermal conductivity	(kg/s)	U	Velocity component in x-direction	(m/s)
$\dot{m}$	Ice melt mass flow rate	($W/m^2.K$)	V	Velocity component in y-direction	(m/s)
n	Normal direction		α	Thermal diffusivity	(m^2/s)
Nu_x	Local Nusselt number		ν	Kinematic viscosity	(m^2/s)
$\overline{Nu}$	Average Nusselt number		θ	Left baffle inclination angle	(degree)
q''	Constant heat flux	(W/m^2)	β	Right baffle inclination angle	(degree)

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